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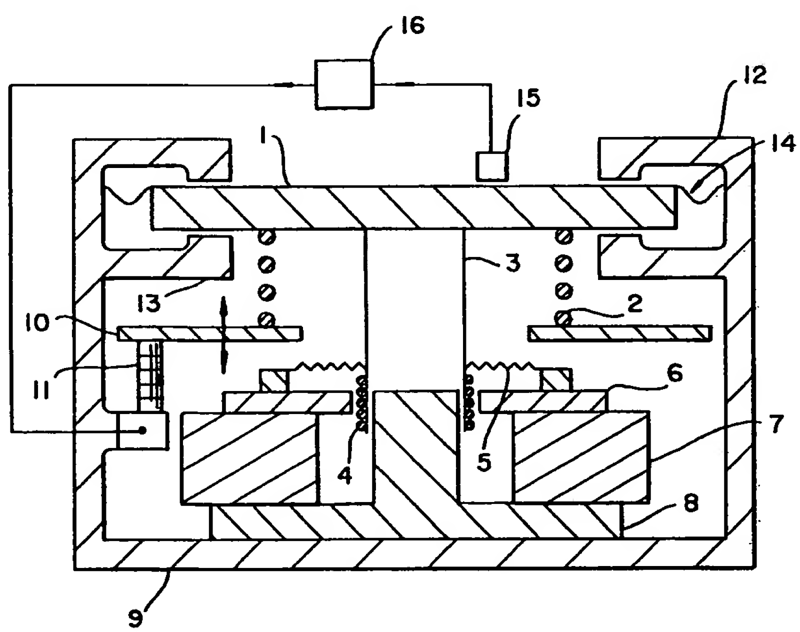
(54) Title: VACUUM SPEAKER



(57) Abstract

A closed box loudspeaker system (9) which has the back of the loudspeaker sealed by a small closed chamber containing a gas which acts like mechanical spring connected to the loudspeaker diaphragm (1). The system avoids the problems inherent in large back chambers by altering the static pressure in the back chamber. To counterbalance the differential pressures either a spring flexible diaphragm (2) or bellows (21) is employed.

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VACUUM SPEAKER

This invention relates to a loudspeaker system having a rear chamber with a static pressure lower than that on the opposite side of the diaphragm.

Conventional loudspeaker systems, for example, sealed box types, employ a large rear chamber to prevent sound from being radiated by the rear side of the loudspeaker diaphragm. This situation often results in inconveniently large enclosures being necessary

Accordingly it is an object of the present invention to eliminate all of the above disadvantage.

It is a further object of this invention to provide an extremely small rear enclosure for a speaker.

A still further object of this is to provide a lower static pressure in the rear chamber of a speaker system that reduces the resonance frequency of the enclosures thereby allowing for a small enclosure.

Another object of this invention is to provide a bellows or spring to counteract the static force on the front of the diaphragm produced by a low static pressure behind the diaphragm.

General Principle of Operation

In a "closed-box" (sometimes called an infinite-baffle) loudspeaker system, the back of the loudspeaker unit is sealed by a closed chamber of volume V in order to stop sound being radiated by the back of the loudspeaker's diaphragm. The gas in the closed box acts like a mechanical spring of stiffness $K = S_D^2 \rho c^2/V$ connected to the loudspeaker diaphragm, where

S_D	is the area of the loudspeaker diaphragm
ρ	s the density of the gas in the closed chamber behind the unit
c	is the speed of sound in the gas of the closed chamber
V	is the closed chamber volume.

Consequently, if other suspension stiffnesses are negligible in comparison to the closed chamber's stiffness, there will be a mechanical resonance of the loudspeaker system at a frequency close to

$$f=\frac{1}{2\pi}\left(S_D^2\,\rho c^2/M_M V\right)^{1/2}$$

where M_M is the total "moving mass" of the loudspeaker diaphragm assembly.

Somewhat above the resonance frequency, f , the output of the loudspeaker will be more or less constant when a constant level voltage drive is applied, the so-called "mass-controlled" region of the response. Somewhat below the resonance frequency, under the same circumstances, the output of the loudspeaker will fall off at 12dB per octave as the frequency of the drive signal is reduced, the so-called "stiffness-controlled" region of the response.

Because of the reduction in performance of the loudspeaker below the resonance frequency, it is usual to operate the loudspeaker above a frequency that is not much different from the resonance frequency. Therefore, in any application where good low-frequency performance is required, the resonance frequency must also be kept low. For a loudspeaker unit of given characteristics, the normal way of reducing the resonance frequency is to make the back-chamber volume, V, bigger, thereby reducing the stiffness of this enclosure. Unfortunately, it frequently happens that to achieve the desired extended low-frequency response of the loudspeaker system, the volume of the back-chamber must be made inconveniently large.

One way of reducing the back-chamber stiffness, $S_D^2\,\rho c^2 / V$, without altering V, is to reduce the value of ρ. This can be readily achieved by reducing the static pressure in the back-chamber to a value p_1, say, that is less than the atmospheric pressure, p. If we assume that the temperature of the gas in the back chamber does not change as the pressure is reduced, then from the gas laws

$$p/\rho = p_1/\rho_1$$

where ρ_1 is the density of the gas in the back-chamber at pressure p_1. Therefore, the stiffness of the back-chamber (K_1, say) becomes

$$K_1 = (p_1/p)\,K$$

which is less than K.

The most important consequence of this is that there is a static force of magnitude $(p-p_1)\,S_D$ exerted on the diaphragm trying to push the diaphragm into the box. This force has to be counterbalanced in some way to enable the loudspeaker to operate. Methods for doing this are described as follows.

From a slightly different viewpoint, suppose that a conventional closed-box loudspeaker system has been designed, and it is desired to reduce the volume of the back-chamber without altering the frequency response of the system. In this case it follows that if the new enclosure volume is V_1, compared to the original volume V, then the enclosure pressures in these two cases must be related by

$$V_1 = (p_1/p)V,$$

so that the back-chamber volume can be reduced by the same factor as the static pressure in that chamber is reduced.

Embodiments of the invention will now be described by way of example with reference to the accompanying drawings, in which

Fig. 1 shows a cross section of a system where a spring is used to counterbalance the static pressure differential,

Fig. 2 shows a cross-section of a system utilizing a second diaphragm to counterbalance the static pressure differential of the system, and

Fig. 3 shows an alternative bellows that can be used to counterbalance the static pressure differential.

One way of counterbalancing the static force on the diaphragm due to the reduction in pressure of the back-chamber is to use a spring. The static straining of the spring provides the counterbalancing force, and the dynamic stiffness of the spring adds to the stiffness of the back-chamber at reduced pressure to give the desired overall suspension stiffness for the loudspeaker diaphragm. To be useful in this application, the spring must be a type that is capable of large static strains whilst maintaining an acceptably small dynamic stiffness.

Referring to Figure 1, there is shown one embodiment of the invention in which a coil spring is used to provide the counterbalancing force. The loudspeaker diaphragm 1 is flat, and constructed of a honeycomb-cored laminated structure of a type well-known in the field of building loudspeakers. There are in addition the usual components of a moving-coil loudspeaker unit such as voice-coil former 3, voice-coil 4, spider 5, top-plate 6, magnet 7, back plate and center pole 8, chassis 9, and surround 14. In addition, there are the coil spring 2, adjustable plate 10, adjusting means 11, and diaphragm stops 12 and 13. The spider, 5, keeps the voice-coil centered, but other arrangements, such as a linear bearing can also be used.

The cavity formed by the sealed chassis 9 (or by an alternative back-chamber if the chassis is not a sealed unit) is held at a pressure lower than atmospheric, and the

consequent static force due to the pressure difference is counterbalanced by the spring 2 which is compressed by the required amount. Because atmospheric pressure changes, it is desirable to mount the coil spring against a movable backplate 10 that can be adjusted by, for example, a motorized screw 11. In this case, the static displacement of the diaphragm that would otherwise occur when the external pressure changes, can be neutralized by adjusting the position of the backplate 10. It is particularly advantageous to have a sensor 15, that measures the average position of the diaphragm, and a feedback control system 16 that adjusts the position of the backplate 10 automatically to maintain the desired average diaphragm position as atmospheric pressure changes. On occasions when the control system is switched off, mechanical stops 12 and 13 can be used to constrain the diaphragm motion to prevent damage. It is also possible to adjust the average diaphragm position by controlling the pressure in the back-chamber where this is preferred to having an adjustable backplate for the spring.

Starting with a standard closed-box loudspeaker design of the desired performance of back-chamber volume V, suppose that one wishes to reduce the back-chamber volume to V_1 without altering the performance. In that case, one has in the notation used previously,

$$K = K_1 + K_s$$

where K_s is the dynamic stiffness of the spring. To provide a balance of the static forces,

$$S_D (p - p_1) = K_s x$$

where x is the required static displacement of the spring, and this assumes that the static and dynamic stiffnesses are not too different. One can show that

$$x = \frac{V(1 - p_1/p)}{S_D \gamma \, [1 - (p_1 V / p V_1)]}$$

As an example, if the original volume, V, was 20 liters; the diaphragm area, $S_D = 0.06 m^2$; γ 1.4; the target new volume $V_1 = 10$ liters and $p_1 / p = 0.1$; then x = 0.216m, and $K_s = 26.3$kN/m.

The spring can be designed so as to minimize its part of the overall moving mass and to avoid resonances that tend to make its dynamic stiffness different from its static stiffness.

Figure 2 shows another embodiment of the invention. In this embodiment, the static counterbalancing force on the diaphragm is supplied by a second, smaller, back-chamber 17 that is pressurized above atmospheric pressure. One wall of the second chamber is a diaphragm 18 that is flexible enough to deform under the dynamic displacements of the loudspeaker diaphragm, and yet strong enough to withstand the static forces on it due to the pressure differences. The diaphragm 18 is connected to the loudspeaker diaphragm 1 through a lightweight link 19. The diaphragm is shown as a cone of the normal type, although this construction is not fundamental to the operation - any diaphragm strong enough to withstand the static pressure forces would suffice.

The second chamber, 17 can be mounted as in the previous embodiment on a backplate that can be moved relative to the magnet assembly, 6, 7 and 8, to allow adjustment for diaphragm offsets due to changes in atmospheric pressure. Alternatively, as shown in Figure 2, the pressure in the second chamber 17 can be adjusted by exchanging gas with a reservoir 20. In Figure 2 a control system comprising a sensor 15 that can be used to derive the average position of the diaphragm 1, and a control means 16 controlling a pump 21 to effect the change in pressure in the second chamber 17, is also shown. As before, the purpose of this control system is to maintain the average position of the diaphragm 1 in the presence of atmospheric pressure changes (or other perturbing influences such as temperature differentials). The pressure in the second chamber can also be controlled by regulating the temperature of the gas with, for example, an electrically heated coil. In this case, the pump and reservoir are not required, and the second chamber can be completely sealed.

Alternatively, the pressure in the back-chamber behind the diaphragm 1 can be adjusted to maintain the average diaphragm position by means that are essentially the same as those indicated for adjusting the second-chamber 17 pressure.

This embodiment relies upon the fact that whilst the static forces on a diaphragm bounding a closed volume are proportional to the area of the diaphragm, the dynamic stiffness of a cavity is proportional to the *square* of the area of the diaphragm. Suppose for example, that the volume of the back-chamber behind the diaphragm 1 is V_1, and the pressure is p_1. Suppose also that the second chamber 17 has a volume of V_2, and its diaphragm 18 has an area of S_s. In that case, the

pressure in the second chamber, p_2, that is required to counterbalance the static pressure forces on the diaphragm is determined by

$$S_s(p_2-p_1)=S_D(p-p_1)$$

The total dynamic stiffness of the suspension of the loudspeaker diaphragm (ignoring spider 5, surround 14, and diaphragm 18 stiffness) is

$$K_T=(S_D-S_s)^2\rho c_1^2/V_1+S^2\rho_2/V_2$$

where ρ_2 and c_2 are the density and speed of sound of the gas in the second chamber 17.

As an example, suppose

S_D	=	0.015	m2
S_s	=	0.0015	m2
V_1	=	10	liters
V_2	=	4	liters
p_1	=	0.1	atmospheres

and that the gas in both chambers is air with an external atmospheric pressure of 1.013×10^5 N/m^2. In that case. the p_2 must be 9.22×10^5 N/m^2, and K_T = 984.4 N/m. This is equivalent to a back-chamber at atmospheric pressure of volume 32 liters.

The diaphragm 18 must be designed to deform over the range of movement of the diaphragm (say $\pm$ 6mm) whilst still being able to withstand the pressure difference p_2-p_1 across it. A diaphragm with a high membrane strength and stiffness, with relatively low bending stiffness is appropriate. A moulded fibre reinforced rubber component is suitable. Similar design considerations apply to the surround 14.

There is an advantage to using gases with low values of γ in either or both of the chambers behind the diaphragm. In particular, if the volume of the second chamber is small enough (so that its fractional change in volume as the loudspeaker diaphragm displaces is significant), the linearity of the suspension stiffness will be improved by using a gas with a low value of γ in this chamber.

An alternative embodiment of the second chamber is a chamber constructed from a flexible bellows. Figure 3 shows such a bellows arrangement, which would

replace second chamber 17 and diaphragm 18 in Figure 2. The same methods of controlling the average diaphragm position can be applied in this case. Rigid end cap 22 takes the place of flexible diaphragm 18 in Figure 2, and the axial displacement of the end cap 22 is accommodated by distortion of the bellow's wall 23. Bellows units are well known for their ability to deform flexibility in the axial direction whilst containing gas at high pressure. The design requires low axial stiffness coupled with the ability to compress or stretch by a few millimeters as the loudspeaker diaphragm moves. At the same time, the bellows must withstand the pressure differential (p_2 - p_1) across its walls. A material that is not susceptible to fatigue (such as phosphor - bronze) is preferred.

Thus, it is seen that by creating a pressure differential to effectively reduce the static pressure behind a speaker diaphragm to a vacuum, a small sealed unit can be provided which eliminates most of the problems inherent in large rear enclosure speakers. The salient features of this arrangement are the partial vacuum and the counter balancing means to offset the pressure differential.

Other embodiments not disclosed, will suggest themselves to those of ordinary skill in the art and are deemed to be covered by this invention which is not limited to the embodiments shown and discussed.

Claims

1. In a loudspeaker system comprising
a first loudspeaker diaphragm means have front and rear faces
mounting means on which said first loudspeaker diaphragm means is mounted so as to be able to vibrate
actuator means in said mounting means adapted to vibrate said first diaphragm means so as to produce sound
sealed chamber means in said mounting means adapted to produce a static pressure differential between said first diaphragm front and rear faces
and force means adapted to counterbalance the effect of said static pressure differential on said first loudspeaker diaphragm whereby said is able to deform over a range of movement necessary to produce sound while simultaneously withstanding the static pressure differential across it.

2. A loudspeaker system as in claim 1 wherein said force means is mounted in said sealed chamber means and exerts force on the rear face of said first diaphragm means.

3. A loudspeaker system as in claim 2 and including adjusting means adapted to vary the force in said force means.

4. A loudspeaker system as in claim 2 wherein including sensing means adapted to measure the average position of the first loudspeaker diaphragm means and a control means adapted to vary the force in said force means in response to signals from said sensing means.

5. A loudspeaker system as in claim 2 wherein said force means comprises a spring means.

6. A loudspeaker system as in claim 2 wherein said force means comprises a second diaphragm means.

7. A loudspeaker system as in claim 2 wherein force means includes a bellows means.

8. A loudspeaker system as in claim 1 wherein said force means comprises a spring means.

9. A loudspeaker system as in claim 8 wherein said force means includes adjusting means adapted to vary the tension in said spring means.

10. A loudspeaker system as in claim 1 wherein said force means comprises a second diaphragm means.

11. A loudspeaker system as in claim 1 wherein said force means comprises a bellows means.

12. A loudspeaker system as in claim 1 wherein said sealed chamber means includes an enclosure formed of said mounting means and voice-coil means adapted to move said first diaphragm means.

13. A loudspeaker system as in claim 1 wherein said force means comprises a second diaphragm means positioned so as to counteract the effect of said static pressure differential, said second diaphragm means containing gas at a predetermined pressure and means connecting said first and second diaphragms.

14. A loudspeaker system as in claim 13 and including a reservoir and gas exchange means adapted to vary the pressure of the gas in said second diaphragm means.

15. A loudspeaker system as in claim 14 included a sensing and control means adapted to control that pressure of the gas in said second diaphragm means in response to signals from said sensing means.

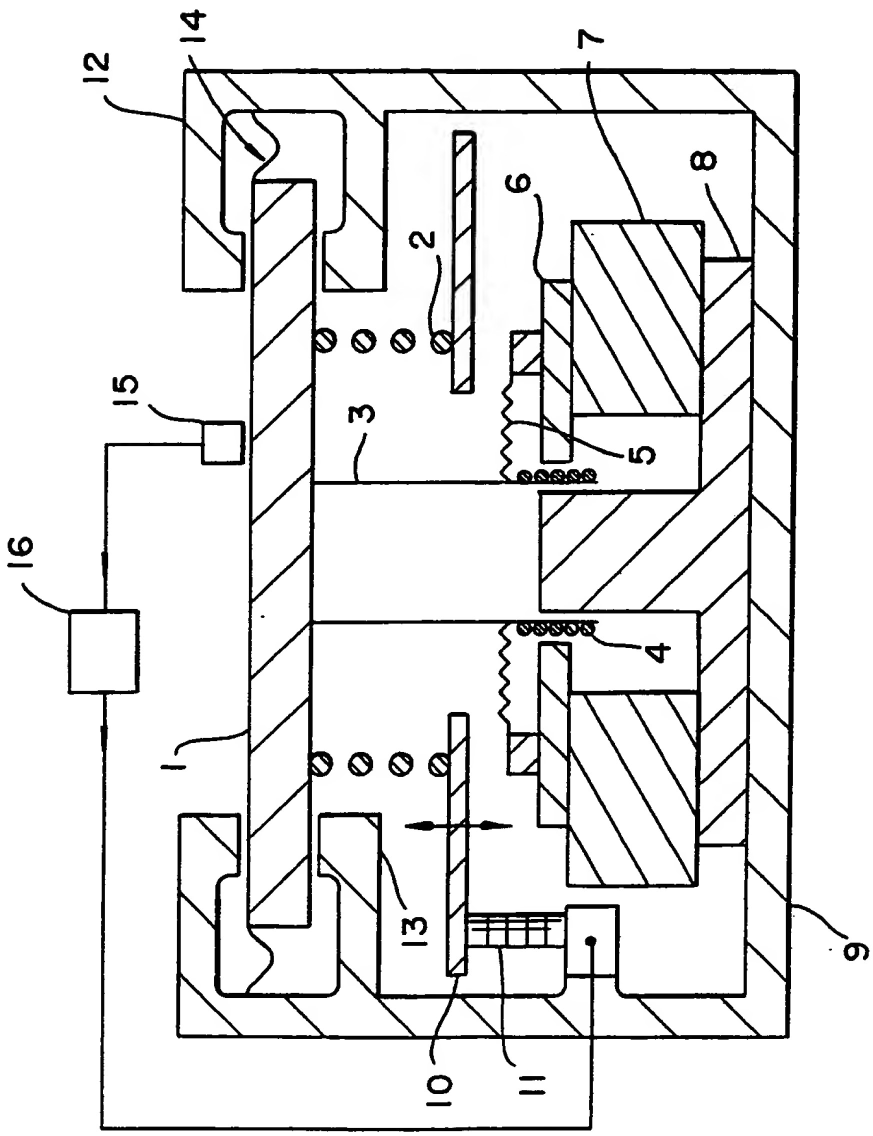

Fig.1

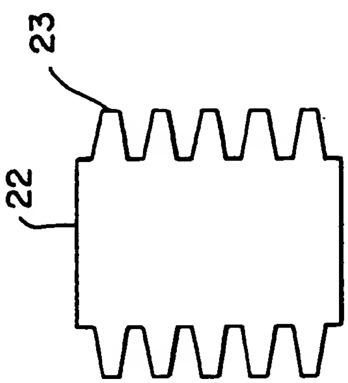

Fig.3

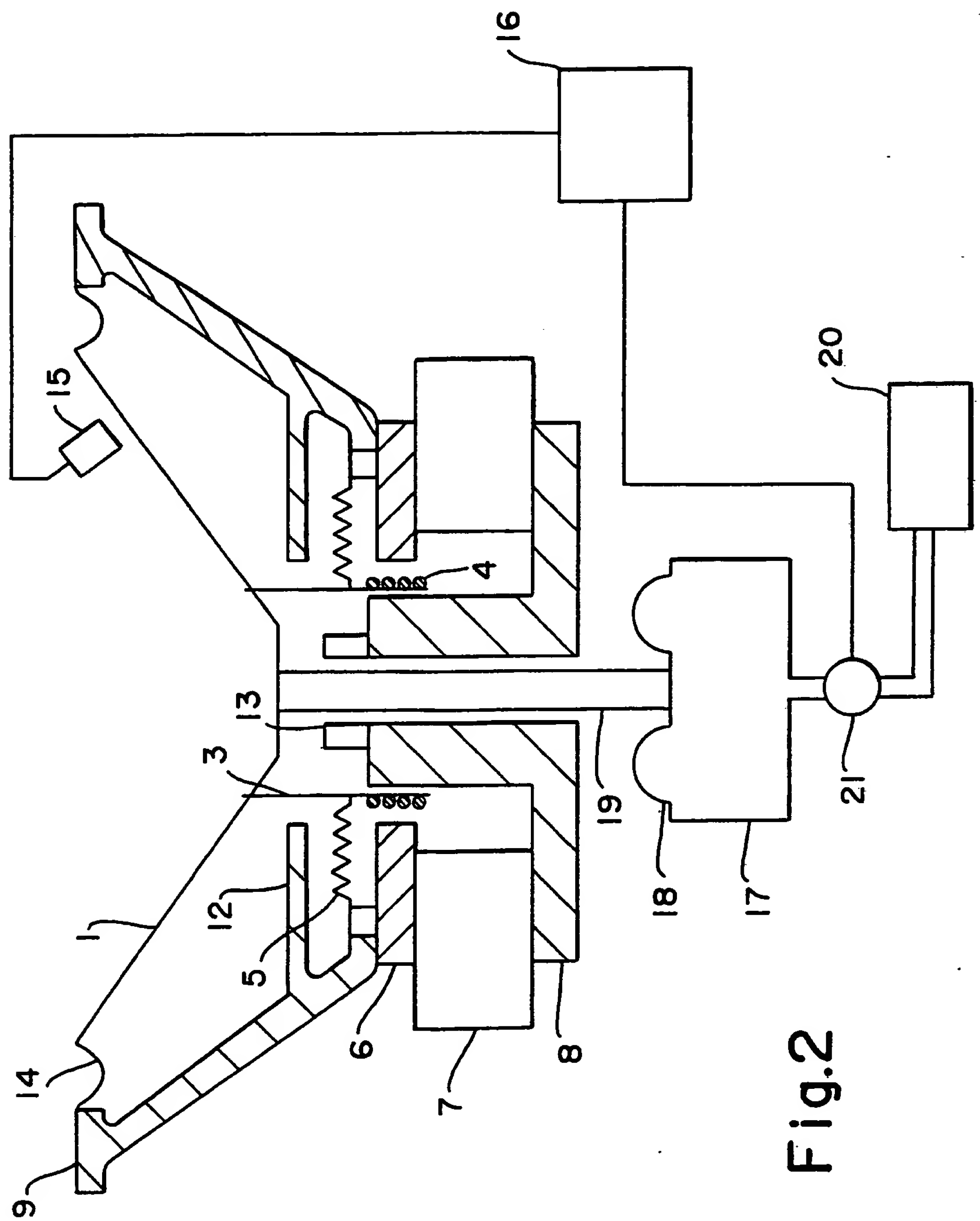

Fig.2

INTERNATIONAL SEARCH REPORT

International Application No. PCT/US91/07324

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) [6]

According to International Patent Classification (IPC) or to both National Classification and IPC

IPC(5): H04R 25/00
U.S. CL.: 381/161, 162, 166

II. FIELDS SEARCHED

Minimum Documentation Searched [7]

Classification System	Classification Symbols
U.S.	381/166, 162, 165, 161, 192, 202, 96 181/161

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [8]

III. DOCUMENTS CONSIDERED TO BE RELEVANT [9]

Category *	Citation of Document, [11] with indication, where appropriate, of the relevant passages [12]	Relevant to Claim No. [13]
X Y	US, A, 4,607,382 (DIJKSTRA ET AL) 19 August 1986 See Figure 1b See Figure 5 See Figure 7	1-13 14,15
A	US, A, 3,772,466 (HOSSBACH) 13 November 1973	1-15
A	US, A, 4,547,631 (NIELIWENDIJK ET AL) 15 October 1985	1-15
A	US, A, 4,595,801 (COFFIN) 17 June, 1986	1-15
A,P	US, A, 5,060,274 (ASAMI ET AL) 22 October 1991	1-15

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IV. CERTIFICATION

Date of the Actual Completion of the International Search	Date of Mailing of this International Search Report
16 March 1992	15 APR 1992
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